EFFECT OF THE DEGREE OF SUPERHEAT ON THE ECONOMY OF A MARSH BOILER-FEED PUMP

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AN INVESTIGATION

OF

THE EFFECT OF THE DEGREE OF SUPERHEAT ON THE ECONOMY OF A MARSH BOILER-FEED PUMP

A THESIS

PRESENTED BY

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EDWIN HOWARD STILLMAN

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Preface

The subject matter of this thesis is divided into three main divisions:

Part I states briefly the object of these tests, and includes a complete description of the apparatus used, an explanation of the operation of the pump and the Fisher hydraulic governor, and the method of calibrating the instruments.

Part II includes a description of the method pursued in the performance of the tests, a discussion of the results, the conclusions drawn, together with curves, tables and original data.

(a) Tests with Saturated Steam.

(b) Tests with Superheated Steam.

Part III is devoted to the bibliography. Few of the articles herein given, apply directly to tests with superheated steam, as very little material on this subject could be found. The few articles which appear to be only remotely connected with the subject were included

because they contain much valuable information relative to tests and operation of pumps.

The Appendix contains the sample calculations.

The application of superheated steam to pumps has been very limited. Very little investigation has been done along this line and consequently little or nothing has been written on the subject. More elaborate tests had been planned, but owing to practical difficulties, many of which have been overcome, the scope of the work has been limited.

In the preparation of the report, especial attention has been given to the description of apparatus, and the operation of the pump and the hydraulic governor. Sketches and photographs have been added where they appeared to add clearness. Only such tables and curves were prepared as seemed necessary to supply the desired information. The sample indicator cards shown are not intended to represent

the average performance of the pump; they were included in order to explain more clearly the cycle of operation in the two cylinders.

We wish to express our obligations to Professor G. F. Gebhardt, professor of Mechanical Engineering at A. I. T. for many valuable suggestions. To Mr. E. S. Libby, instructor in Experimental Engineering, A. I. T., we owe our indebtedness for the use of much auxiliary apparatus. To Mr. J. C. Peebles, instructor in Mechanical Engineering at A. I. T., we are indebted for his assistance in the calibration of the thermometers, especially for the use of the platinum resistance pyrometer and the high temperature standard. For the calibration curve of the venturi meter we owe our thanks to Messrs. S. J. Aurelius and J. S. Harvey, A. I. T., '09. To two of our classmates, Messrs. F. C. Lanzig and M. Vanderkloot, Jr., we express our gratitude for the cheerful assistance which they rendered during our tests. We are also very much

indebted to Mrs. Julia A. Beveridge, librarian at A. I. T., for her kind assistance in making up what little of the bibliography we herein are able to present.

Jesse I. Menkin.

Edwin H. Stillman.

Chicago, May 25, 1909.

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Part I.

Part II.

Part III.

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Illustrations.

Part I.

Part II.

Part I.

Apparatus.

Calibration of Instruments.

Part I.

Object

The purpose of these tests was to determine the effect of the degreee of superheat on the economy of a 200-gallon Marsh boiler-feed pump.

Apparatus

The general layout of apparatus is shown in the sketch on page 1(a). Water was taken from a reservoir or sump, below the floor level, and discharged into weighing tanks or thru a venturi meter. Steam was supplied to the pump either from the main steam line, or by-passed thru the superheater and thence turned into the pump. The exhaust steam was led into a surface condenser and the condensed steam was pumped into the weighing tanks shown.

The Pump

The pump is shown in the photograph on page


ENT of APPARATUS
SH PUMP TESTS
TUTE of TECHNOLOGY
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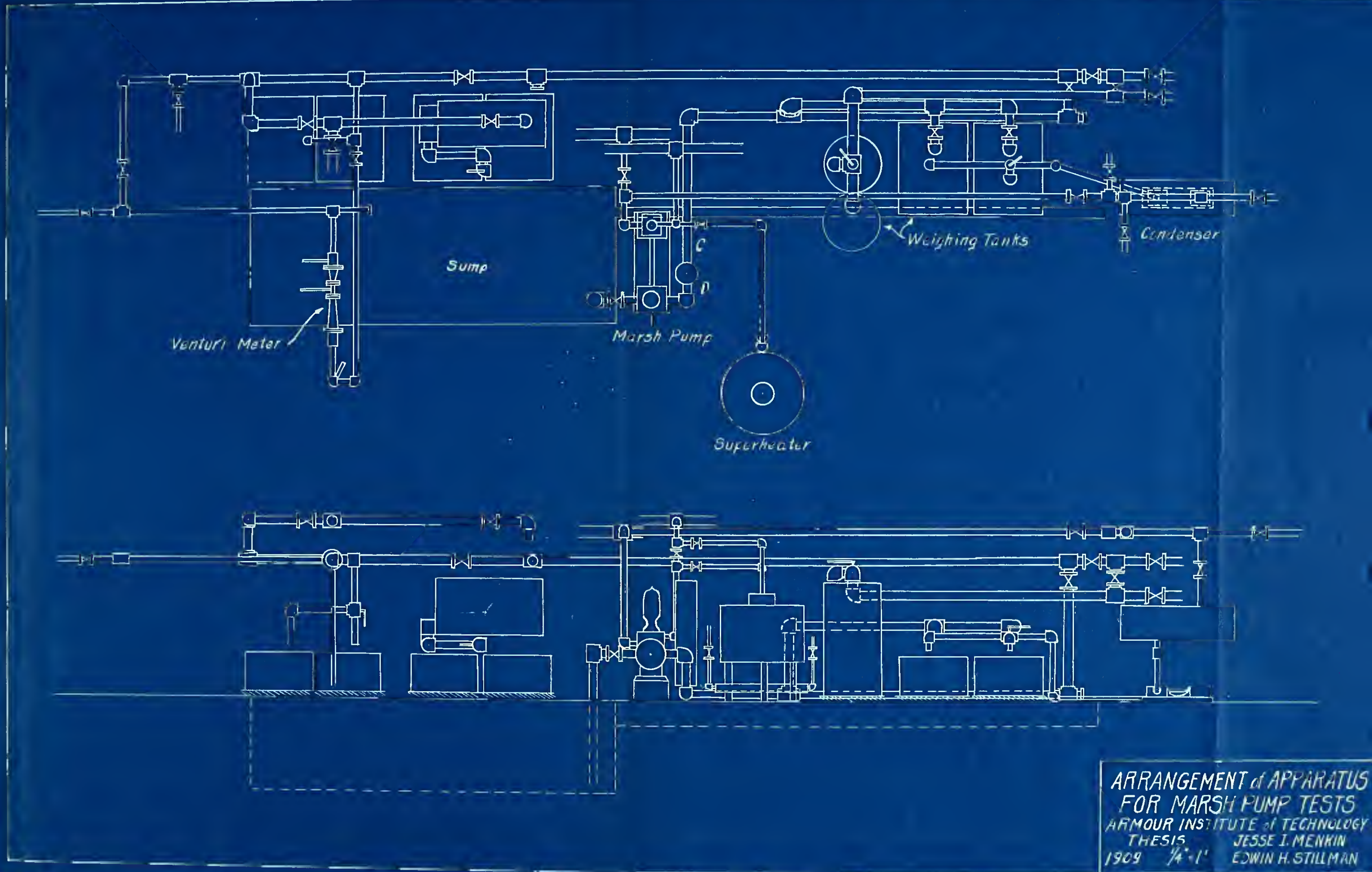
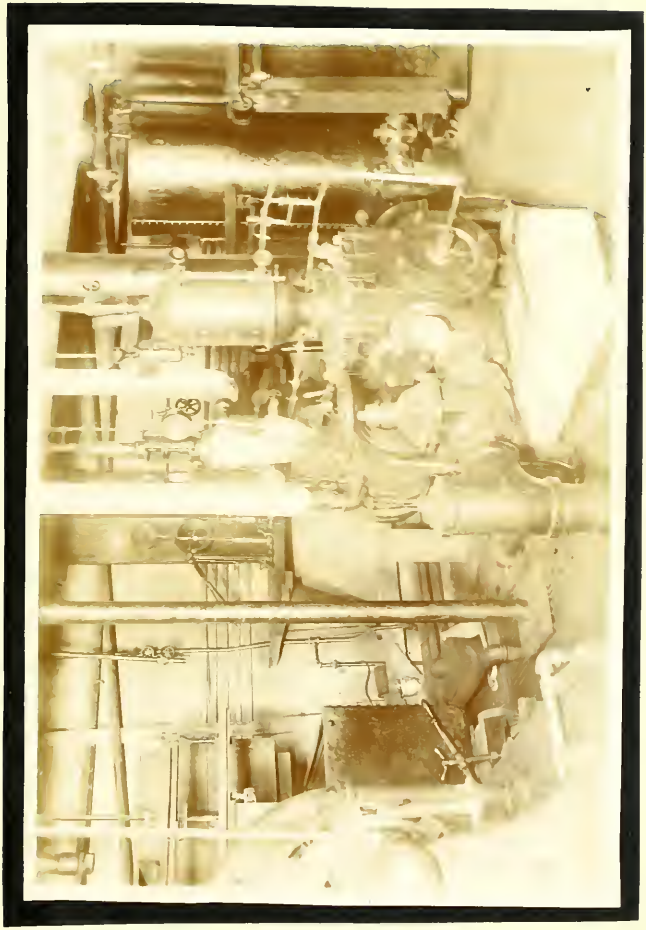
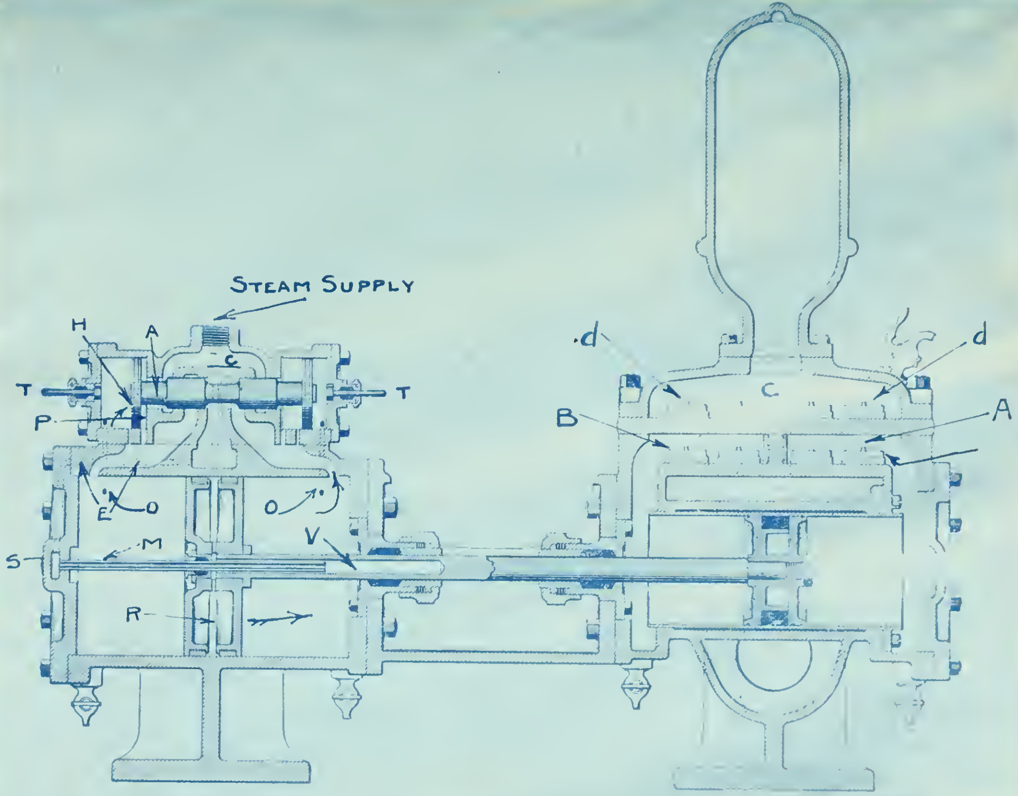

Sump
Venturi Meter
Marsh Pump
C
D
Weighing Tanks
Condenser
Superheater
ARRANGEMENT of APPARATUS
FOR MARSH PUMP TESTS
ARMOUR INSTITUTE of TECHNOLOGY
THESIS
JESSE I. MENKIN
1909
1/4"=1'
EDWIN H. STILLMAN



1 (b)

1(b), which was taken before connection to the superheater. It is direct acting, having a steam cylinder 12" in diameter, a water cylinder 7-1/4" in diameter, and a maximum stroke of 12". Its rated capacity is 216 gallons per minute at a speed of 100 strokes per minute when operating with a steam pressure of 80 pounds per square inch gauge against a discharge pressure of 125 pounds per square inch.

The cylinders are separate castings, each cylinder and base or support forming a unit, the two units being connected as shown in figure on page 2(a). The steam piston is equipped with two metallic packing rings; the water plunger, with hydraulic packing. The two are connected by a brass rod 1-3/8" in diameter. To the head of the steam cylinder is attached a hollow brass tube 11/16" in diameter, which serves as a passage for the steam entering the hollow piston. A portion of the piston rod is hollow-



SECTIONAL VIEW
MARSH STEAM PUMP

2 (a)

ed out so that the tube has a sliding fit. From the head end of the water plunger an 11/16" brass tail rod extends thru the cylinder head, and connects to the reducing mechanism. The air dome is connected to a stand pipe, this in turn being connected to a Westinghouse air compressor. In these tests, however, the air compressor was not required.

Valve Gear

Referring to figure on page 2(a), steam enters the chest C, and passing thru the annular opening A, formed between the reduced neck of the valve and the chest pass is projected against the inner surface of the valve head H, before escaping into the cylinder by means of port P. Both the pressure of the steam and the impulse due to its velocity in striking the valve head H, force the valve to the left in the direction of the current, thus tending to close or re-

strict the annular steam passage A. As the steam reaches the cylinder, the piston is driven to the right. Steam from the cylinder entering port C, flows upward into the valve chest and exerts a counter pressure on the left side of the valve head H, tending to drive it to the right - a movement which would give greater port opening to the entering chest from the chest C. The valve, therefore, is always balanced, and occupies a position depending upon the relative intensity of the two forces which tend to move it in opposite direction -- <u>Admission Steam</u> -- which tends to close it, and <u>Cylinder Steam</u>, which tends to open it wider. This constitutes the steam governing element.

The steam piston consist, as shown, of a spool form, each head being provided with a metallic packing ring, the interior space R forming a reservoir for live steam which is supplied from the upper chamber of the steam

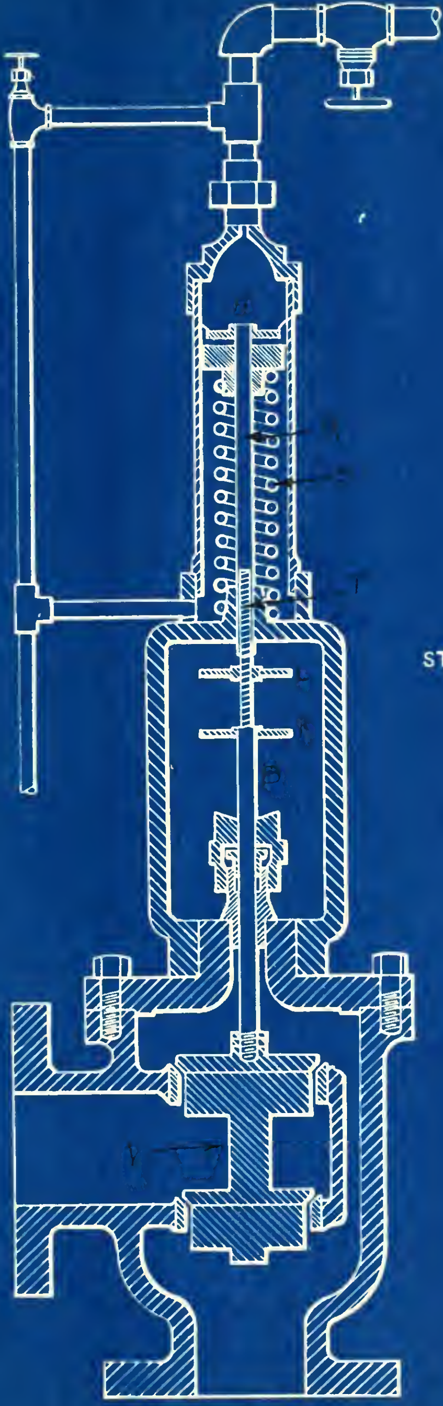
chest by passage E to the cylinder cap S, thence by tube K and the hollow piston V. The pressure of this steam is used only for "tripping" or reversing the valve by admitting steam alternately against the outer surfaces of the valve heads H thru the connecting passages C,C, near each end of the cylinder. The tappets T are used for moving the valve by hand in case it fails to start automatically.

Water Cylinder

The water cylinder, page 2(a), is divided into two compartments A and B, head and crank ends. As the plunger moves to the left, water is drawn thru the disc valves A, A, into the head end: at the same time water is forced from the crank end thru the valves c, c into Chamber C. The movement of the plunger to the right delivers the head end charge and refills the crank end.

The Hydraulic Governor

The sketch, page 6(a) shows the Fisher hydraulic governor. The spring S tends to hold the double-balanced valve V off its seat thus allowing admission steam at full pressure. The pressue of the water in the delivery pipe, transmitted thru pipe P acts on the piston tending to overcome the resistance of the spring and close the valve. Hence it is seen that with given steam and discharge pressure, the valve will assume some balanced position, and that any rise or fall in delivery pressure will tend to close or open the valve respectively. If a lower delivery head is desired, the resistance of the spring can be adjusted by means of hand wheel K and lock wheel L, so that the valve will assume a position of rest with a lower water pressure at A. The piston rod R is pinned to the sleeve T and the valve stem B which is rigidly attached to wheel K, is screwed into this sleeve, thereby altering the



FISHER
STEAM PUMP GOVERNOR

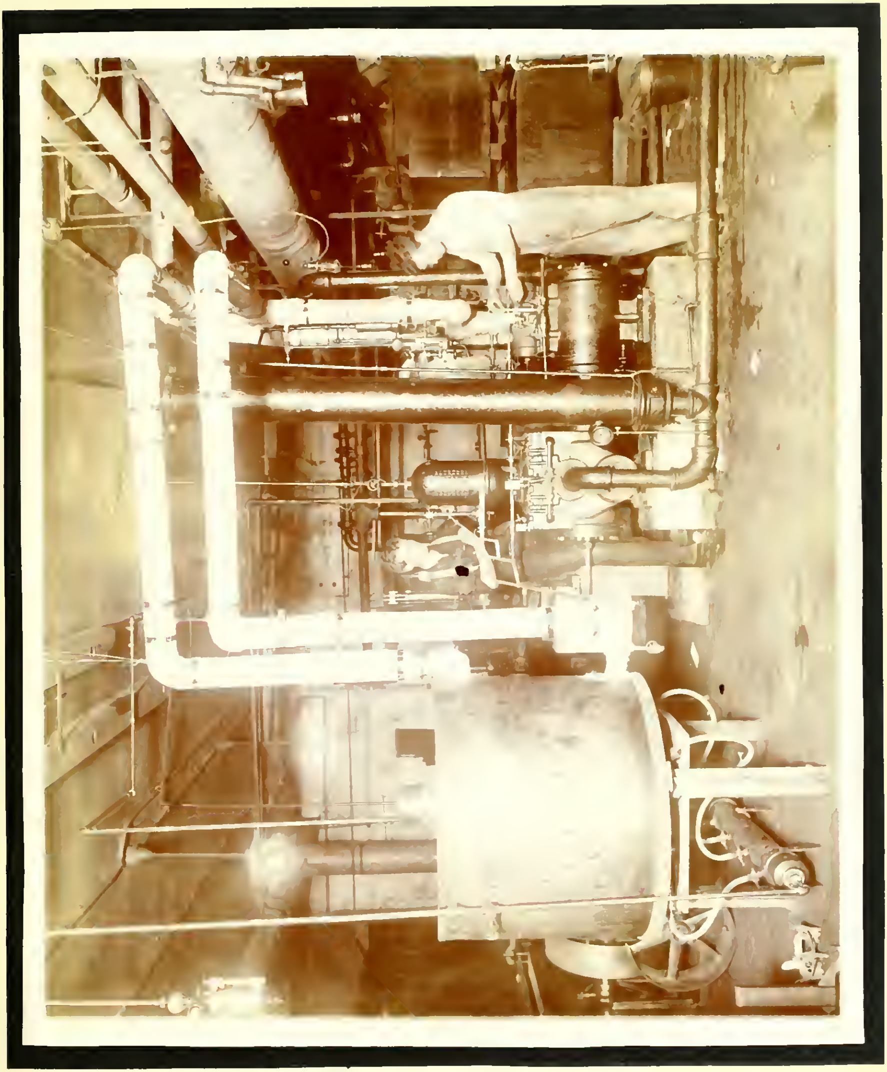
tension of the spring.

Reducing Motion

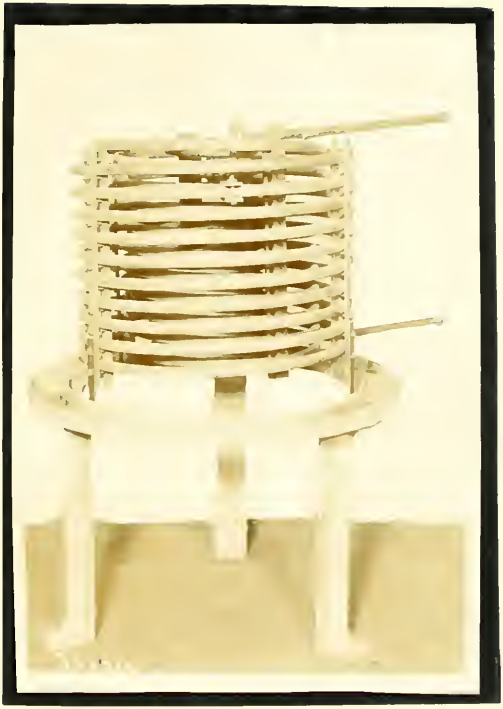
The reducing mechanism is shown in the photograph on page 7(a). To the tail rod is attached a pantograph which transmits motion to a 5/8" steel rod mounted in suitable bearings, so that its motion is exactly parallel to that of the piston rod. Mounted on this rod at points equidistant from the indicators are two aluminum "fingers" to which are attached indicator cords made equal in length and as short as possible. Thus both indicators are actuated by the same mechanism without any appreciable error due to unequal stretch of the cords due to a difference in their lengths.

The Superheater

The superheater consists of three concentric coils of 3/4" steel pipe, 10", 20", and 28"

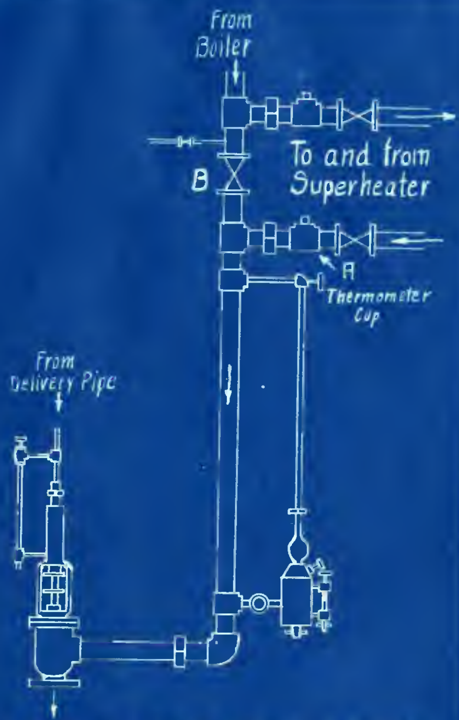


7 (a)



7 (r)

in diameter, each of 10-1/2 turns. The three coils are welded together making 160' of continuous pipe. Steam enters at the top, passes downward in the outer coil, thence upward thru the central coil, downward thru the innermost coil and out at the bottom, to the pump. The detail of the steam piping showing the connections to the superheater is shown on page 8 a). The spaces between the turns in the coils are partially filled with strips of asbestos which serve to deflect the heated gases and afford a uniform distribution of heat. The coils are surrounded by two sheet iron shells, 30" and 36" in diameter. The interveing space is filled with cinders to within 1" of the top, to prevent radiation. At this point, around the circumference of the inner shell is a series of 1/2" holes which allow the products of combustion to escape between the shells to the stack at top of cover. The coils are mounted on a three-legged circular base about 24" in height. Heat is supplied by 4 - 40 cubic foot



DETAIL of STEAM PIPING at PUMP
Showing Connections with
Fisher Governor
and
Superheater

gas burners feeding from a 4" x 3' reservoir connected to two 3/5" gas mains.

Condenser

The condenser is a Wainright surface condenser consisting of 113 tubes 5/8" external, 1/2" internal diameter and 54.5" in length, having a steam surface of 84 square feet and a water surface of 67.3 square feet. A 1/2" opening permits venting to the atmosphere when condensing at atmospheric pressure. A small Marsh pump delivers the condensed steam to the weighing tanks.

Piping

The main steam pipe is 1-1/2" in diameter. The by-pass to the superheater is 1-1/2" and is connected to the superheater by reducing ells. The exhaust steam pipe is 2" in diameter. The suction pipe is 4" in diameter and is fitted with

a gate valve and a thermometer cup for determining the temperature of the water.

The discharge pipe is 4" at the pump and reduces to 2" at the tanks. To the right of the stand pipe — photograph, page 7(a) — a 1/2" pipe from the discharge line transmits the water pressure to operate the governor. The stand pipe is 8" in diameter and 8' high. A 1/2" pipe connects it and the air dome of the pump, to a Westinghouse air compressure so that any desired pressure can be obtained. During these tests the compressor was not used.

Calibration of Instruments

The thermometers used were all Fahrenheit and were calibrated against an 800° nitrogen borosilicate standard. This standard was in turn calibrated with a platinum resistance pyrometer equipped with a Whipple temperature indicator — #1930 — made by the Cambridge Scien-

tific Instrument Company. The nitrogen standard was found to be correct. The thermometers were inserted in a portable blast furnace — see photograph page 11(a) — which was heated by a gas flame.

The venturi meter had been previously calibrated by Messrs. Aurelius and Harvey, M. I.T., '09, who plotted a calibration curve which was used in these tests — see thesis of S. J. Aurelius and J. S. Harvey — "Comparison of Efflux Coefficients for various shapes and sizes of Nozzles" —.

The pressure gauges were calibrated with a Crosby dead-weight tester; the steam gauge was found to read 5 pounds too high; the discharge gauge was correct. The vacuum gauge was calibrated with a Wheeler surface condenser and read correctly.

11(a)

The Tabor indicator springs were calibrated in the steam line against a calibrated steam gauge. Calibration of the springs were made before the saturated runs, and again before the superheated runs. The true scales are indicated on the log sheets.

Part II.

Method.

Tests with Saturated Steam.

Tests with Superheated Steam.

Part II.

Method

The general method was to send saturated steam thru the superheater where it was raised to the desired temperature, thence turned into the pump. The temperature of the saturated steam was determined from the gauge pressure at the throttle; the temperature of super - heated steam was determined by means of a thermometer in steam pipe just before entrance to the governor — see sketch, page 8(a) — .

A series of preliminary tests were made with saturated steam during which great difficulty was experienced in taking indicator cards due to excessive pounding in the water cylinder. Especially was this true at high rates of speed. It was also found that the weighing tanks were of insufficient capacity to handle the quantity of water discharged.

The slip was also excessive being in the neighborhood of 10 per cent. It was therefore deemed advisable to overhaul the pump. Accordingly, it was dismantled, the valves examined, and new metallic packing rings were made for it. A new spring ring of 1/8" steel was made for the water plunger and the hydraulic packing was renewed. To reduce the pounding, a 1/4" valve was placed in the suction line. The quantity of air thus drawn in with the water was reduced to a minimum so that the consequent decrease in capacity was negligible. Owing to the poor condition of the pump the results of these tests were not kept.

It was originally planned to make a series of performance tests at varying speeds with saturated steam, and a series of comparative tests at varying speeds with varying degrees of superheat. After completing the first

superheated tests it was found that time would not permit carrying out the original plan. It required from 2-1/2 to 5 hours to get the temperature of the superheated steam constant. It was, therefore, decided to make the tests with superheated steam at constant speed, varying only in the degrees of superheat.

Tests With Saturated Steam

These tests were of 30 minutes duration. Observations were taken at intervals of 10 minutes, of steam, exhaust and discharge pressures, temperatures in the calorimeter, suction water and room. The steam pressure was determined by means of a steam gauge, the exhaust pressure was measured by a vacuum gauge, and the discharge pressure by a pressure gauge mounted in the discharge pipe near the pump. The steam supply and speed were regulated by means of the Fisher governor; the discharge pressure was maintained constant at 60 pounds per square inch by means of a valve in the discharge line. The suction head was measured with a rule at the start and finish of each run, the average being used in the calculations. The quality of steam was determined with a throttling calorimeter; the temperature of the water with a thermometer

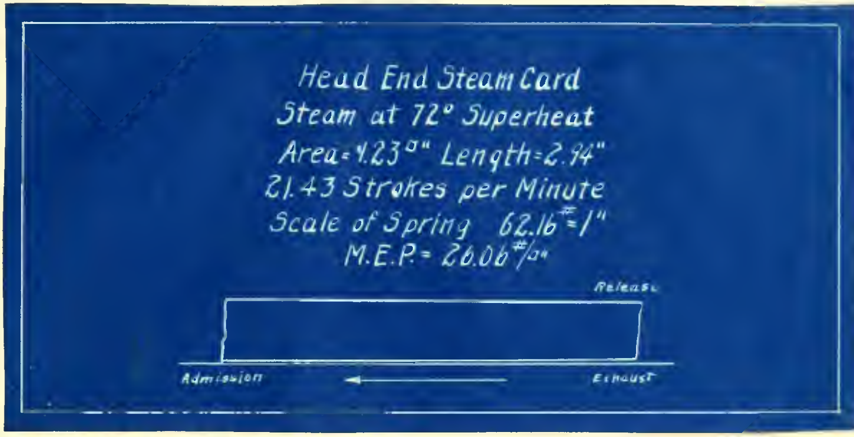
inserted in a suitable oil cup placed in the suction pipe close to the pump. The exhaust steam was discharged into a Wainright surface condenser condensing at atmospheric pressure. The steam condensed was pumped into weighing tanks and weighed once during each run. No account was taken of the temperature nor the weight of cooling water used in the condenser.

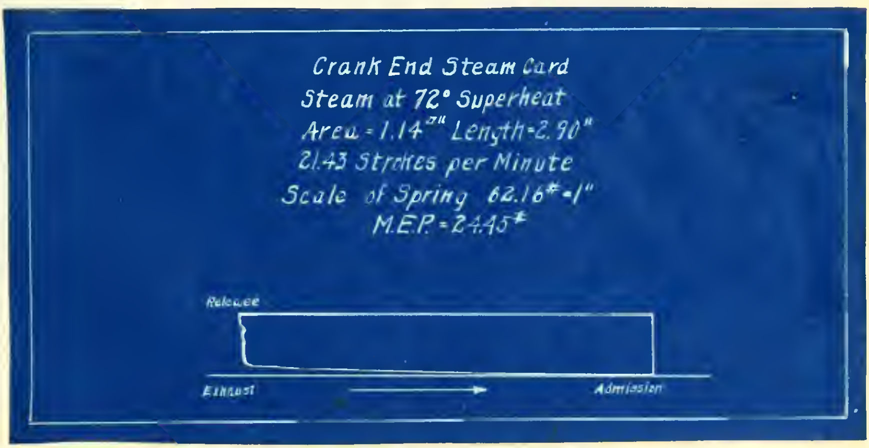
The water pumped was delivered to tanks and weighed. When the speed exceeded 40 strokes per minute, it was found impracticable to weigh the water, so that the venturi meter was used. When the capacity became too great for the mercury column to handle, a portion of the water from the pump was bypassed to the weighing tanks; the total water pumped, being the sum of weighed and metered water.

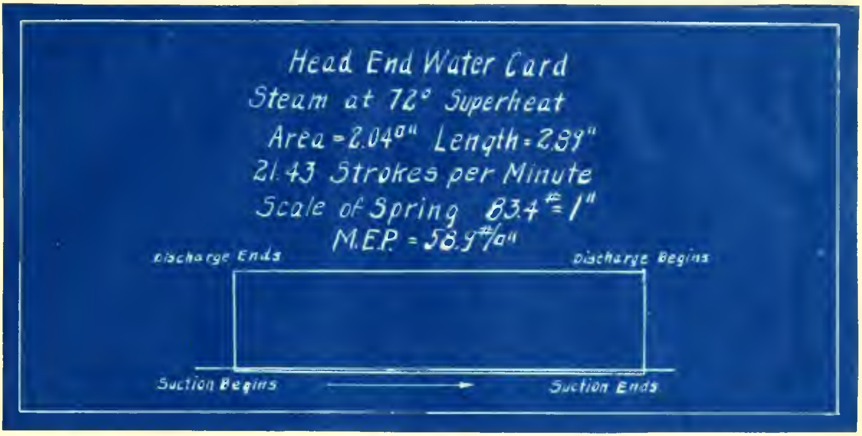
Indicator cards were taken simultaneously from both cylinders every ten minutes, individual cards being taken for head and crank ends, 18(a), 18(b).

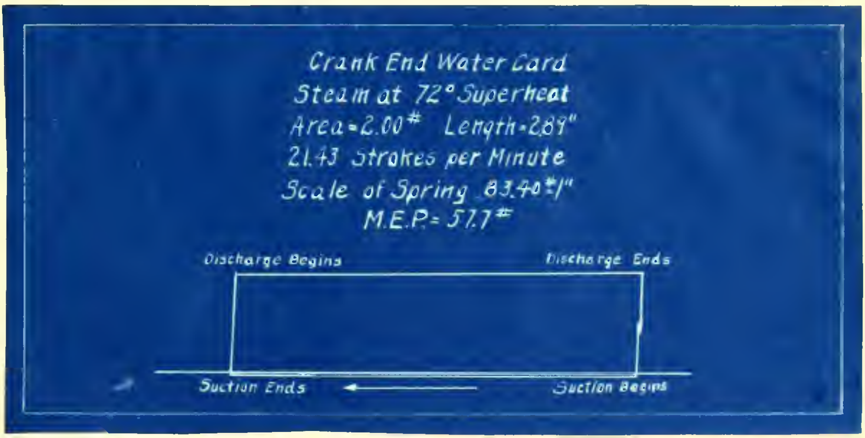
The speed of the pump was found by means of a stroke counter attached to the tail rod, which registered double strokes. This was read at the start and finish of each run.

The ratio of reduction was determined by means of a pencil, mounted on the tail rod, which traced a line on a sheet of paper while another line was simultaneously drawn by the indicator pencil. The ratio of the length of the piston stroke to the length of the line drawn on the indicator card, on the same stroke, is the ratio of reduction. This ratio was determined on the crank end stroke of the water cylinder. A great many determinations were made at various speeds. There was con-


Head End Steam Card
Steam at 72° Superheat
Area=1.23□" Length=2.94"
21.43 Strokes per Minute
Scale of Spring 62.16#=1"
M.E.P.=26.06#/□"
Release
Admission
Exhaust


Crank End Steam Card
Steam at 72° Superheat
Area=1.14□" Length=2.90"
21.43 Strokes per Minute
Scale of Spring 62.16#=1"
M.E.P.=24.45#
Release
Exhaust
Admission


Head End Water Card
Steam at 72° Superheat
Area = 2.04□" Length = 2.89"
21.43 Strokes per Minute
Scale of Spring 83.4# = 1"
M.E.P. = 58.9#/□"
Discharge Ends
Discharge Begins
Suction Begins
Suction Ends

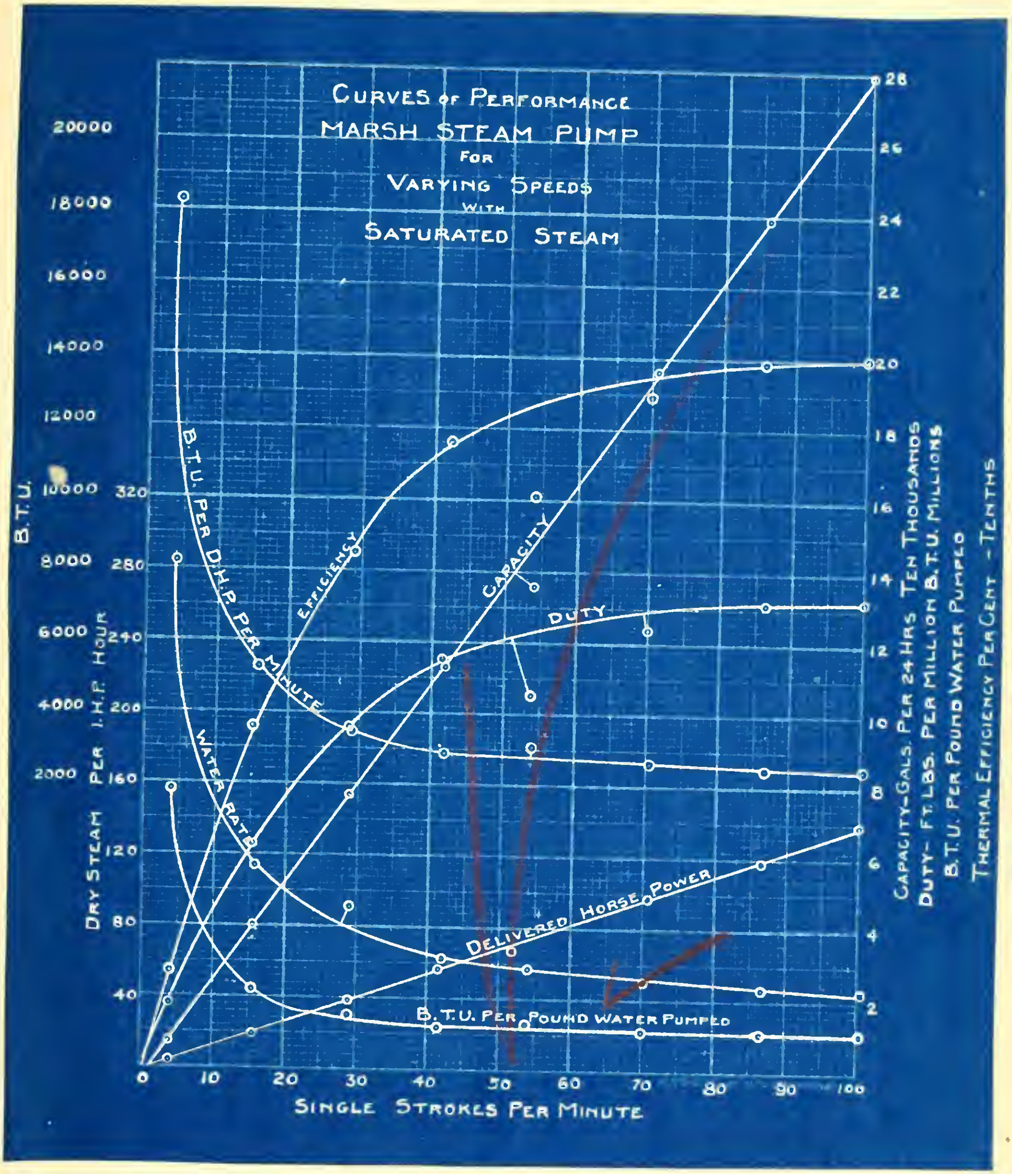

Crank End Water Card
Steam at 72° Superheat
Area = 2.00# Length = 2.89"
21.43 Strokes per Minute
Scale of Spring 83.40#/1"
M.E.P. = 57.7#
Discharge Begins
Discharge Ends
Suction Ends
Suction Begins

siderable variation, but it followed no general law. The average of these was taken as the probable true ratio. This variation is probably due to imperfections in the reducing motion or to a slight bending of some of its links.

Discussion

The results of these tests are shown graphically by the curves on page 19(a). With the exception of the test at 54 strokes per minute, the points fall quite uniformly on the curves. The error in this run is paobably due to the inaccuracy of the venturi meter, at low heads, this being the first run in which the meter was used.

The curves show that the capacity and horse power increase directly with the speed.



Curves of Performance
Marsh Steam Pump
For
Varying Speeds
With
Saturated Steam
B.T.U.
20000
18000
16000
14000
12000
10000
8000
6000
4000
2000
Dry Steam Per I.H.P. Hour
320
280
240
200
160
120
80
40
0
10
20
30
40
50
60
70
80
90
100
Single Strokes Per Minute
28
26
24
22
20
18
16
14
12
10
8
6
4
2
Capacity-Gals. Per 24 Hrs Ten Thousands
Duty- Ft. Lbs. Per Million B.T.U. Millions
B.T.U. Per Pound Water Pumped
Thermal Efficiency Per Cent - Tenths
Efficiency
Capacity
Duty
B.T.U. Per D.H.P. Per Minute
Water Rate
Delivered Horse Power
B.T.U. Per Pound Water Pumped

The efficiency and duty increase with the speed, reaching a maximum at about 100 strokes per minute. The general trend of the curves indicate that, if the pump could be operated at a speed exceeding 100 strokes per minute, there would be no increase in efficiency or economy. It is also evident that the economy increases greatly with the speed. From the water rate curve, the steam consumption at 4 strokes per minute is 284 pounds per I. H. P. hour, at 20 strokes it has already decreased to ~~204~~ 100, while at 40 strokes it is as low as ~~148~~ 64. From there on, the decrease is comparatively slight reaching a minimum of ~~168~~ 44 pounds at 100 strokes per minute.

The B. T. U. curve shows exactly the same characteristics. At the slowest speed — 4 strokes per minute — 18000 B. T. U. are supplied per delivered horse power per minute. At 20 strokes this has already

decreased to 4400, and at 40 strokes only 3000 B. T. U. are required. At 100 strokes the heat consumption is only 2400 B. T. U. Thus it is seen that an increase of 20 per cent in speed at low velocities increases the economy $\frac{(1800-4400)}{1800} = 75.5$ per cent.

In conclusion, we may say that when operating with saturated steam:

1. The minimum efficient speed is 40 strokes per minute.
2. That the economy increases with the speed.
3. That the capacity, duty, horse power, and efficiency increase with the speed.
4. That the maximum efficiency, duty and capacity are obtained at the rated speed of 100 strokes per minute.

Tests with Superheated Steam.

At the commencement of these tests it was found that the drop in pressure thru the superheater was nearly 50 per cent of the maximum steam pressure attainable with saturated steam. In the former tests a maximum steam pressure, at the higher speeds, of 80 pounds gauge could be obtained; with steam thru the superheater, at low degrees of superheat, the steam pressure dropped to 45 pounds and the maximum attainable speed was 36 strokes per minute. The speed throughout these tests was maintained approximately constant, at about 22 strokes per minute. The discharge pressure was kept constant at 60 pounds per square inch. The temperature of the superheated steam was kept as nearly constant as possible throughout the run.

At low degrees of superheat not much difficulty was encountered in keeping the degree of superheat constant throughout the run. At

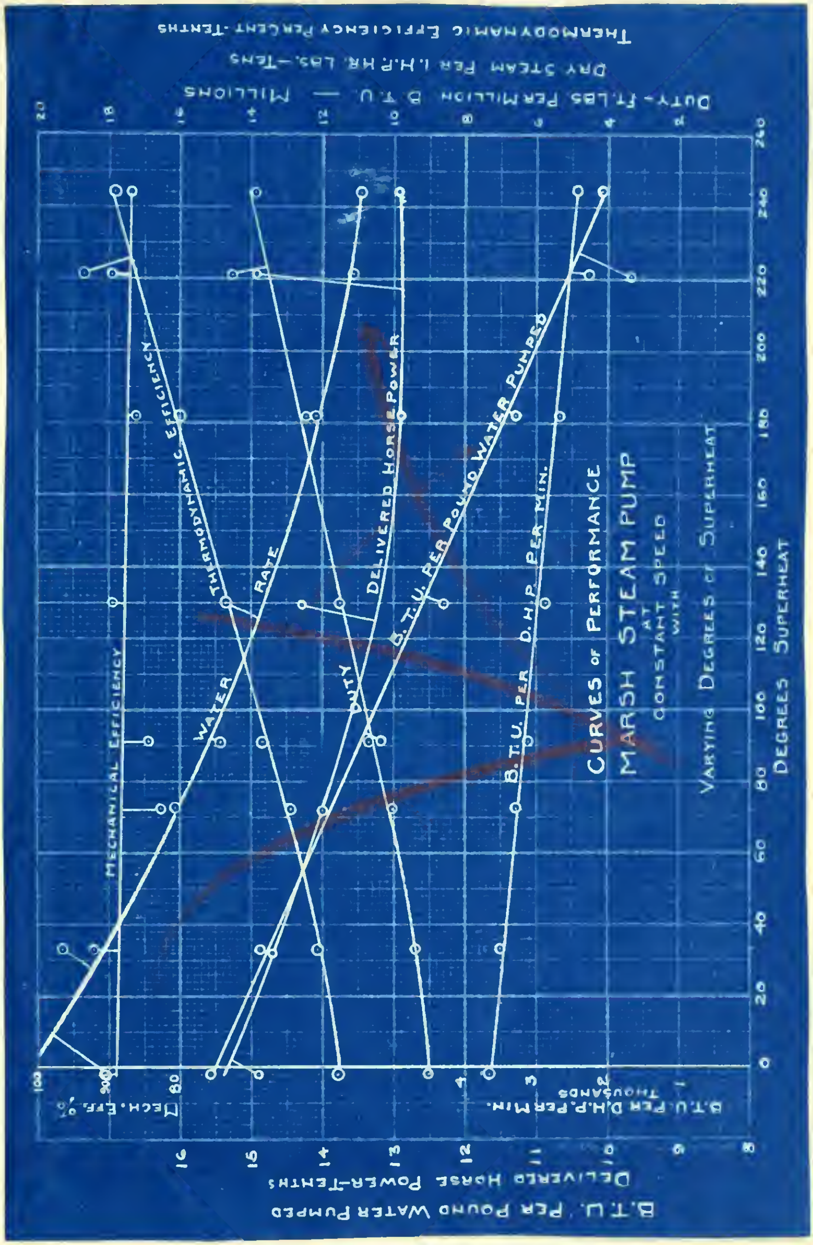
high degrees of superheat there was a fluctuation of 5 or 6 degrees, and more time was required to attain even this condition. The adjustment of the steam governor had to be made with more care, as it became much more sensitive as the temperature range increased. The discharge valve likewise had to be watched more carefully as there was considerable fluctuation in the pressure.

Temperatures of superheated steam were read every 2 minutes, the average value being recorded for the run. Observations of the steam and discharge pressures, suction head, and temperature of water were recorded every 10 minutes. The speed counter was read at the start and finish, individual indicator cards were taken, as in the tests with saturated steam.

Discussion

From the results obtained, the curves, page

24(a) were drawn. The delivered or pump horse power is seen to have a general decrease during the series of superheated runs. This, however, cannot be attributed to the increase of superheat, as the delivered horse power is a function of speed of pump, length of stroke, and delivery head. But it is probably due to the inability to keep the speed for the various runs exactly constant. The high and low points seem to correspond with the high and low speeds respectively. There is likewise a constant decrease in the mechanical efficiency with increase in the degree of superheat. This is probably on account of incresed friction due to the unequal expansion of the piston and cylinder walls. To obviate this, the cylinder should be lagged to prevent so great a difference of temperature betwen the inner and outer walls. The water rate and heat supplied, decrease quite rapidly with an increase of superheat. The duty and thermodynamic


CURVES OF PERFORMANCE
MARSH STEAM PUMP
AT
CONSTANT SPEED
WITH
VARYING DEGREES OF SUPERHEAT
DEGREES SUPERHEAT
DUTY - FT. LBS PER MILLION B.T.U — MILLIONS
DRY STEAM PER I.H.P. HR. LBS.—TENS
THERMODYNAMIC EFFICIENCY PER CENT.-TENTHS
B.T.U. PER POUND WATER PUMPED
DELIVERED HORSE POWER—TENTHS
B.T.U. PER D.H.P. PER MIN.
THOUSANDS
MECH. EFF. %
MECHANICAL EFFICIENCY
THERMODYNAMIC EFFICIENCY
WATER RATE
DUTY
DELIVERED HORSE POWER
B.T.U. PER POUND WATER PUMPED
B.T.U. PER D.H.P. PER MIN.


CONSTANT SPEED

Average Log
Test of
Marsh Steam Pump
Condensing at Atmospheric Pressure

At Varying Speeds with Saturated Steam — At Constant Speed with Superheated Steam

Item No.	No. of Run	1	2	3	4	5	6	7	8	0	1	2	3	4	5	6	7
	Date	4/19/09	4/19/09	4/19/09	4/19/09	4/19/09	4/19/09	4/19/09	4/19/09	4/27/09	4/30/09	4/30/09	5/6/09	4/30/09	5/6/09	5/7/09	5/7/09
1	Degree Superheat	—	—	—	—	—	—	—	—	0	33	72	90.8	129.3	181.4	220.8	244.1
2	Duration of Run Mins.	28	30	30	30	30	30	30	30	30	30	30	30	30	30	30	30
3	Single Strokes per Min.	3.96	15.86	29.2	41.73	53.93	70.3	86.73	100.66	22.88	22.17	21.43	25.07	21.93	20.5	22.77	21.33
4	Length of Stroke Ft *	0.82	0.903	0.895	0.889	0.914	0.953	0.975	0.978	.885	.900	.895	.851	.890	.880	.872	.850
5	Pressures — Barometer #/□"	14.43	14.43	14.43	14.43	14.43	14.43	14.43	14.43	14.35	14.44	14.44	14.22	14.44	14.22	14.42	14.42
6	Pressures — Steam — Gage #/□"	32.5	35	37.5	37.5	39.5	44.5	43.7	36.3	34.35	34.1	33.35	33.7	32.35	30.25	31.25	29.6
7	Pressures — Steam — Absol #/□"	46.93	49.43	51.93	51.93	53.93	58.93	58.13	50.73	48.10	48.54	47.79	47.92	46.79	44.47	45.67	44.02
8	Pressures — Exhaust #/□" Abs.	14.43	14.43	14.43	14.43	14.43	14.43	14.43	14.43	14.35	14.44	14.44	14.22	14.44	14.22	14.42	14.42
9	Pressures — Discharge #/□"	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60	60
10	Pressures — Venturi Meter P-P, Ins. Hg.	—	—	—	6.44	9.3	19.35	23.18	19.11	—	—	—	—	—	—	—	—
11	Temperatures — Calorimeter	216	216	213	218.5	216	225	224	228	216.7	—	—	—	—	—	—	—
12	Temperatures — Saturated Steam	—	—	—	—	—	—	—	—	278.3	279.0	278.0	278.2	276.7	273.6	275.2	272.9
13	Temperatures — Superheated Steam	—	—	—	—	—	—	—	—	—	312	350	369	406	455	496	517
14	Temperatures — Boiling Temp. Atm. Press.	211.1	211.1	211.1	211.1	211.1	211.1	211.1	211.1	210.7	211	211	210.3	211	210.3	211	211
15	Temperatures — Water	62	64.3	67.8	76	82	85.3	90	95.7	69	70	67	70	65	64	64	63
16	Temperatures — Room	77	79	78.5	81	77.5	76	85	89	73	80	80	91	80	93	72	72
17	Head, Feet of Water — Suction	2.1	2.1	2.0	2.0	2.0	2.0	2.0	2.0	2.0	2.1	2.0	2.1	2.1	2.0	2.0	2.1
18	Head, Feet of Water — Discharge	138.5	138.5	138.5	138.5	138.5	138.5	138.5	138.5	138.5	138.5	138.5	138.5	138.5	138.5	138.5	138.5
19	Head, Feet of Water — Dist. Betw. Gages	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0	0
20	Head, Feet of Water — Total	140.6	140.6	140.5	140.5	140.5	140.5	140.5	140.5	140.5	140.6	140.5	140.6	140.6	140.5	140.5	140.6
21	Steam — Condensed — Total Lbs.	84	136	171	208	272	333	384	445	167	153	135	138	117	95	92	83
22	Steam — Condensed — Per Hr. Lbs.	180	272	342	416	544	666	768	890	334	306	270	276	234	190	184	166
23	Steam — Condensed — Per I.H.P. Hr. Lbs.	290	116.5	95.1	63.8	58	51.8	47.2	45.9	185.4	193	161.7	147.8	146.5	121.7	111.1	109
24	Steam — Dry — Total Lbs.	82.5	133.4	167.1	204.05	266.1	326	377	439	164	153	135	138	117	95	92	83
25	Steam — Dry — Per Hr. Lbs.	176.8	266.7	334.2	408.1	532.3	652	754	879	328	306	270	276	234	190	184	166
26	Steam — Dry — Per I.H.P. Hr. Lbs.	284.5	114.7	92.9	62.7	56.8	50.6	46.3	45.3	182.1	193	161.7	147.8	146.5	121.7	111.1	109
27	Steam — Dry — Per Pound Water Pumped Lbs.	.068	.0193	.0128	.0106	.0115	.0098	.0093	.0092	.0313	.0148	.0137	.0128	.0116	.0105	.0088	.0092
28	Steam — Dry — Per D.H.P. Hr. Lbs.	942	268.5	176.6	147.2	160	135	128	128	220	208	193	179	163	147	123.5	128.5
29	Steam — Dry — Per D.H.P. Min. Lbs.	15.6	4.48	2.93	2.45	2.67	2.25	2.13	2.13	3.67	3.46	3.22	2.98	2.72	2.46	2.06	2.15
30	Steam — Quality %	98.24	98.05	97.73	98.1	97.85	98.2	98.16	98.7	98.2	—	—	—	—	—	—	—
31	Steam — Specific Heat C_p	.61	.61	.63	.60	.61	.58	.58	.57	.58	.58	.54	.54	.52	.51	.50	.50
32	Water Pumped — Total — Pounds	1233	7017	13315	19650	23550	34000	41430	48252	10474	10307	9883	10815	10063	9073	10440	9070
33	Water Pumped — Total — Cu. Ft.	19.8	112.5	214	315.9	378.9	546.3	666	778	168	165.5	158.5	173.6	161.5	145.6	167.4	145.4
34	Water Pumped — Total — Gallons	148.1	841.7	1600	2361	2835	4080	4992	5820	1258	1238	1185	1298	1209	1090	1252	1087
35	Water Pumped — Per Hour — Pounds	2644	14034	26630	39300	47100	68000	82860	96504	20948	20614	19766	21630	20126	18146	20880	18140
36	Water Pumped — Per Hour — Cu. Ft.	42.5	225	428	631.8	757.8	1092.6	1332	1556	336	331	317	347.2	323	291.2	334.8	290.8
37	Water Pumped — Per Hour — Gallons	317.8	1683.4	3200	4722	5670	8160	9984	11640	2516	2476	2370	2596	2418	2180	2504	2174
38	Water Pumped — Gallons per Min.	5.3	28	53.3	77	94.4	136	166	193	41.9	41.25	39.5	43.25	40.25	36.35	41.7	36.2
39	Water Pumped — Gallons per 24 Hrs.	7630	40400	76800	113200	136000	196000	239500	279500	60400	59400	57000	62300	58000	52300	60000	52200
40	Plunger Displacem. Cu. Ft./Hr.	54	240.8	440	623	828	1127	1423	1653	340.3	335	322	369.2	328	303.2	334	305
41	Per Cent Slip	21.5	6.57	2.73	—	—	3.16	5.62	5.87	1.28	1.19	1.55	5.95	1.52	4.12	—	4.66
42	Scale of Spring — Steam	62.16	62.16	62.16	62.16	62.16	62.16	62.16	62.16	62.16	62.16	62.16	62.16	62.16	62.16	62.16	62.16
43	Scale of Spring — Water	82.39	82.39	82.39	82.39	82.39	82.39	82.39	82.39	83.4	83.4	83.4	83.4	83.4	83.4	83.4	83.4
44	M.E.P. — Steam	27.3	23.1	19.95	25.15	26.25	27.9	28.1	28.65	25.64	23.3	28.5	25.0	24.0	24.78	24.08	24.3
45	M.E.P. — Water	58.9	60.8	61.3	60.3	62.45	67.35	70	74.05	59.95	59.6	58.75	60.2	59.8	60.8	60.85	59.63
46	I.H.P. — Steam	0.621	2.33	3.60	6.51	9.37	12.89	16.28	19.39	1.800	1.884	1.668	1.869	1.596	1.560	1.655	1.522
47	I.H.P. — Water	0.463	2.13	3.91	5.46	7.53	11.83	14.49	17.84	1.653	1.458	1.380	1.572	1.428	1.342	1.477	1.322
48	Delivered Horse Power	0.188	0.996	1.89	2.79	3.32	4.83	5.89	6.85	1.486	1.468	1.404	1.537	1.430	1.288	1.482	1.287
49	Friction Loss Betw. Cyls. %	25.0	8.58	—	19.25	19.65	14.4	11.0	8.0	8.17	7.95	17.25	15.9	10.52	13.96	10.75	13.12
50	Mechanical Efficiency %	75.0	91.42	—	80.75	80.35	85.6	89.0	92.0	91.83	92.05	82.75	84.1	89.48	86.04	89.25	86.88
51	B.T.U. Supplied — Per Hr.	207,000	313,500	392,000	480,000	625,500	769,500	887,000	1,027,000	326000	308000	277200	286000	247000	205300	202000	184000
52	B.T.U. Supplied — Per I.H.P./Min.	5550	2242	1813	1227	1113	995	909	882	3015	3240	2770	2550	2580	2195	2033	2013
53	B.T.U. Supplied — Per D.H.P./Min.	18340	5250	3452	2867	3140	2656	2512	2495	3650	3600	3290	3100	2880	2660	2270	2400
54	B.T.U. Supplied — Per Pound Water Pumped	78.2	22.33	14.7	12.21	13.29	11.3	10.7	10.63	15.55	14.94	14.02	13.2	12.28	11.30	9.68	10.13
55	Duty per Million B.T.U. Ft. Lbs.	1,795,000	6,300,000	9,560,000	11,800,000	10,580,000	12,420,000	13,110,000	13,200,000	9,040,000	9,410,000	10,040,000	10,650,000	11,450,000	12,430,000	14,540,000	13,900,000
56	Therm. Equiv. of Work Done/Hr.	477	2,540	4,800	7,100	8,500	12,280	14,930	17,400	3785	3740	3575	3915	3642	3280	3775	3277
57	Thermodynamic Eff. %	0.272	0.961	1.45	1.76	1.61	1.89	1.99	2.01	1.16	1.21	1.29	1.37	1.47	1.60	1.87	1.78

* Avge. of Head & Crank Water Cyl.

Diameter of Steam Cylinder 12 Ins.
Diameter of Water Plunger 7½ Ins.
Diameter of Piston Rod 1⅞ Ins.
Diameter of Tail Rod [illegible] Ins.

Area of Piston: Head 113.1 Sq. Ins.; Crank 111.6 Sq. Ins.
Area of Plunger: Head [illegible] Sq. Ins.; Crank [illegible] Sq. Ins.

Ratio of Reduction 3.709

Geo. L. Monkin
Edwin H. Stillman

Running Log

Test of

Marsh Steam Pump

at

Varying Speeds

with

Saturated Steam

Indicators — Water Tabor 82.39# Spg. — Steam Tabor 62.16# Spg.

Apr. 19, 1909

No of Run			1		2		3		4		5		6		7		8
Pressures #/□"	Steam	10:48	38	11:57	40	12:48	42	2:58	43	3:58	44	4:38	48	5:47	50	6:51	43
		58	38	12:07	40	:38	43	3:08	42	4:08	44	:48	48	:57	48	7:01	43
		11:08	37	:17	40	1:08	42	:18	42	:18	45	:58	50	6:07	48	:11	40
		:16	37	:27	40	:16	43	:28	43	28	45	6:08	52	:17	48	:21	39
	Exh.		0		0		0		0		0		0		0		0
			0		0		0		0		0		0		0		0
			0		0		0		0		0		0		0		0
			0		0		0		0		0		0		0		0
	Disch.		60		60		60		60		60		60		60		60
			60		60		60		60		60		60		60		60
			60		60		60		60		60		60		60		60
			60		60		60		60		60		60		60		58
Temperatures	Calorimeter		217		219		215		220		224		228		224		226
			218		217		215		218		224		229		224		230
			217		217		215		218		224		226		224		229
			218		217		215		217		224		220		224		229
	Water		62		64		67		76		90		84		90		95
			62		64		68		76		90		85		90		96
			62		64		68		76		90		86		90		96
			62		65		68		76		90		86		90		96
	Room		76		78		80		81		85		76		85		89
			78		80		75		81		85		76		85		89
Suction Head Ins.		10:48	25	11:57	24.5	12:48	24.5	2:58	23	3:58	24.5	4:38	23.8	5:47	24.5	6:51	24.5
		11:16	25	12.27	25	1:16	24.5	3:28	23.8	4:28	24.5	5:08	23.8	6:17	24.5	7:21	24
Cond. Stm. Lbs.			84		136		171		208		272		333		384		445
Water Pumped #			1233		7017		13315		0		0		0		4235		14552
Stroke* Counter		10:48	683.5	11.57	1023	12:48	548	2:58	207	3:58	5706	4:38	6835	5:47	29474	651	2455
		11:16	739	12:27	1261	1:18	986	3:28	833	4:28	6515	5:08	7890	6:17	30775	721	3965
Venturi Meter Ins. Hg.			—		—		—		6.44		9.3		19.35		23.18		19.11
Barom. Ins Hg			29.4		29.4		29.4		29.4		29.4		29.4		29.4		29.4

* Double Strokes

Running Log
Test of
Marsh Steam Pump
at
Constant Speed
with
Superheated Steam

Indicators	Water Tabor	83.4# Spg.
	Steam Tabor	62.16 Spg.

Date			4/27/09		4/30/09		4/30/09		5/6/09		4/30/09		5/6/09		5/7/09		5/7/09
No of Run			0		1		2		3		4		5		6		7
Pressures #/□"	Steam	4:33	34	2.03	33.	5:40	32.	12:33	32.	8:55	31	5:00	29.5	1:06	29.	3:03	28
		43	34	:13	34	150	33	43	33.5	9:05	31	10	29.5	16	31.	13	28
		53	33	:23	36	6:00	33	53	33.	9:15	32	20	28.5	26	31.	23	28
		5:03	34	2:33	32	6:10	33	1:03	33.5	9:25	33	5.30	29.5	1:36	30.	3:33	30
	Exh.		0		0		0		0		0		0		0		0
			0		0		0		0		0		0		0		0
			0		0		0		0		0		0		0		0
			0		0		0		0		0		0		0		0
	Disch.		60		60		60		60		60		60		60		60
			60		60		60		60		60		60		60		60
			60		60		60		60		60		60		60		60
			60		60		60		60		60		60		60		58
Temperatures	Calorimeter		210		312		350		367		402		451.		490		508
			210		312		349		366		402		451.		490		510
			210		313		349		367		402		450.		490		511
			210		311		348		368		402		452.		490		511
	Water		68		70		70		70		66		64		64		63
			70		70		70		70		65		64		64		63
			69		70		70		70		64		64		64		63
			69		70		70		70		64		64		64		63
	Room		72		80		80		91		80		92		73		71
			74		80		80		91		80		94		71		72
Suction Head Ins.		4:33	25.5	2:03	24.5	5:40	26.	12:33	25	8:55	26.	5:00	25.	1:06	23.5	3:03	26.
		5:03	24.5	2:33	25.	6:10	23.	1.03	25.	9:25	24.5	5:30	24.	1:36	24.	3:33	24.5
Cond. Stm. Lbs.			167		153		135		138.		117		95.		92		83.
Water Pumped #			10474		10307		9883		10815.		10063		9073.		10440		9070.
Stroke* Counter		4.33	535.	2:03	3857.5	5:40	293.	12.33	5914	8:55	478.	5:00	9073	1:06	284.	3:03	171.
		5.03	878	2:33	4190.	6:10	614.5	1.03	6290	9.25	807.	5:10	9380.5	1:36	3152.5	3:33	491.
Barom. Ins. Hg			29.24		29.4		29.4		29.98		29.4		29.98		29.4		29.4

*Reads double strokes.

AVERAGE DATA
from
INDICATOR CARDS
TEST of MARSH STEAM PUMP
1909

SATURATED STEAM—VARIABLE SPEED										
NUMBER of RUN			1	2	3	4	5	6	7	8
STROKES PER MINUTE			3.96	15.85	29.20	41.73	53.93	70.30	86.73	100.66
QUALITY of STEAM (%)			98.24	98.05	97.73	98.10	97.85	98.20	98.16	98.70
STEAM END	HEAD	AREA of CARD (Sq.in)	1.21	1.16	.99	1.20	1.29	1.36	1.49	1.59
		LENGTH of CARD (In)	2.68	2.93	2.91	2.95	3.00	3.12	3.18	3.18
		LENGTH of STROKE (FT)	.827	.906	.900	.912	.928	.965	.982	.982
	CRANK	AREA of CARD	1.19	1.05	.89	1.18	1.25	1.44	1.38	1.35
		LENGTH of CARD	2.78	3.03	2.96	2.93	3.01	3.12	3.16	3.20
		LENGTH of STROKE	.860	.937	.915	.906	.931	.965	.978	.990
WATER END	HEAD	AREA of CARD	1.97	2.15	2.14	2.11	2.23	2.51	2.67	2.83
		LENGTH of CARD	2.63	2.92	2.86	2.87	2.94	3.08	3.15	3.16
		LENGTH of STROKE	.813	.903	.885	.887	.910	.953	.975	.977
	CRANK	AREA of CARD	1.79	2.16	2.17	2.10	2.25	2.53	2.69	2.85
		LENGTH of CARD	2.61	2.92	2.93	2.88	2.97	3.08	3.15	3.16
		LENGTH of STROKE	.806	.903	.905	.890	.918	.953	.975	.978

CONSTANT SPEED-VARIABLE DEGREE of SUPERHEATED STEAM APPLIED										
NUMBER of RUN			0	1	2	3	4	5	6	7
STROKES PER MINUTE			22.88	22.17	21.43	25.07	21.93	20.50	22.77	21.33
DEGREE of SUPERHEAT			0	33.	72.	90.8	129.8	181.4	220.8	244.1
STEAM END	HEAD	AREA of CARD	1.22	1.153	1.24	1.18	1.15	1.208	1.12	1.093
		LENGTH of CARD	2.91	2.95	2.95	2.83	2.92	2.945	2.863	2.783
		LENGTH of STROKE	.899	.911	.911	.874	.904	.909	.884	.860
	CRANK	AREA of CARD	1.18	1.05	1.17	1.10.	1.10	1.14	1.09	1.09
		LENGTH of CARD	2.91	2.93	2.93	2.84	2.92	2.905	2.87	2.808
		LENGTH of STROKE	.899	.905	.905	.876	.904	.897	.886	.867
WATER END	HEAD	AREA of CARD	2.08	2.11	2.04	2.00	2.10	2.07	2.04	1.953
		LENGTH of CARD	2.872	2.91	2.905	2.75	2.893	2.85	2.813	2.748
		LENGTH of STROKE	.887	.899	.897	.850	.893	.880	.868	.850
	CRANK	AREA of CARD	2.04	2.08	2.04	1.98	2.03	2.083	2.08	1.983
		LENGTH of STROKE	2.86	2.92	2.893	2.76	2.873	2.843	2.837	2.755
		LENGTH of STROKE	.883	.901	.893	.852	.887	.878	.875	.850

efficiency increase almost directly with the degree of superheat.

That greater economy is obtained with an increase in the degree of superheat, may be seen from the following table. Starting with saturated steam, or zero degrees of superheat, the gain in per cent of B.T.U. supplied at zero degrees superheat is --see curve B.T.U. per B.H.P. per minute--, noted in following table:

Degree Superheat	Gain in Per Cent
75	11.---$\frac{(3600-3100)}{3600}$ = .11.
100	14 ---$\frac{(3600-3100)}{3600}$ = .14.
150	21 ---$\frac{(3600-2850)}{3600}$ = .21.
200	24 ---$\frac{(3600-2750)}{3600}$ = .24.
240	33 ---$\frac{(3600-2400)}{3600}$ = .33.

The water rate at 0° is 200 pounds per I. H. P. hour. The decrease in the water rate can be seen below:

Degree Superheat	Dry Steam per I.H.P. Hr.
0	200.
75	159.
100	148.
150	130.
200	118.
240	110.

Conclusion

The results of this test would indicate that with the pump operating at constant speed,

1. The economy is a direct function of the degree of superheat.

2. That with the pump in its present condition the mechanical efficiency decreases with an increase in the degree of superheat.

3. That the thermodynamic efficiency increases with the degree of superheat.

From this it does not necessarily follow that the economy of all pumps, nor even this pump, in actual use as a boiler feed, would be increased by the use of superheated steam, as much would depend upon the method employed in superheating the steam. Whether or not the economy would be increased when the cost of producing the superheat is considered, is beyond the scope of this thesis. This will depend entirely upon existing conditions and is a problem that must be left to the operating engineer to decide.

Respectfully submitted,

May 25, 1909.

Jesse I. Menkin

Edwin D. Stillman

Part III.

Bibliography.

Bibliography.

Marsh Boiler Feed Pump.

Engineering.

60:946

Pumping Tests with the Superheated Steam

U. S. Engineer

37:140 Jne. 1, '00.

Pumping Engine Test with Superheated Steam.

Engineering.

70:231 Dec.14,'00.

Pumping Engine with Superheated Steam.

Eng. Record

41:473 May 19,'00.

Pumping Engine Tests with Superheated Steam

Power

20:33 Dec. '00.

Conditions of Max. Pumping Engine Efficiency.

Eng. News

44:212 Sept.27,'00.

Boiler Feed-Pump Efficiency

Engr. London

83:152 Feb. 5,'97.

Feed-Pump Efficiencies.

Elect. World.

36:455 Sept. 7,'01.

Comparative Steam Pump Tests with Saturated Steam.

Eng. News

51:301 Mar. 31,'04.

Tests of Small Steam Pump for Boiler Feed.

Eng. Record

44:233 Sept.7,'01.

Consumption of Steam Pumps.

Am. Loc. Mech. Engr.

17:145 - 95

Determining the Capacity of a Power Pump.

Power

Oct. 27,'99.

Rules & Formulas for Steam Pumps.

Engr. — U. S. A. -

36:49 Mch. 1,'99.

Action of Pumps.

Eng. Mag.

25:123 Apr. '03.

Some Pump Problems.

Am. Mach.

22:79 Feb. 2,'99.

Comparative Value of Different Arrangement of Air chambers on Pumps.

Power

21:11 Jan. '01.

Setting the Valves on Steam Pumps.

Power

17:4	Jan.)
7	Feb.)
7	My.) 1897.
7	Jne.)
9	Nov.)

Power 1.

Power

18:19 Dec. 1898.

Gebhardt — Steam Power Plant Engineering.

p. 408

p.475

Rogers — Pumps & Hydraulics.

J. C. Library 621.6

Q5CC

Economy Test of Feed-Pumps.

Eng. — London —

87:40 Jan. 13,'99.

:118 Feb. 3,'99.

:345 Apr. 7,'99.

Appendix

Calculations

The following series of calculations are for run number 1 of the series of superheated tests (only those being shown which were deemed necessary for explanation) and are presented to show the methods used in securing the results which appear on the Final Log of page 24 b). With the exception of the determination of the quality of steam at admission, these computations show the methods for securing the results of our saturated runs also. Hence, the calculations for the quality of steam of the first saturated run are included in the following. In all cases, the data used, consists of the average corrected observations of the test under consideration. Peabody's tables were used for heat values.

Item 3. Single stroke per minute.

Speed counter reading at start, 3857.5

Speed counter reading at finish, 4190.

$\frac{(4190-3857.5)}{30} \times 2 =$ 22.17 single strokes per min.

Item 4. Length of stroke

Av. length of head end cards of water cyl. = 2.91"

$291 \times \frac{3.704}{12} = 8.99'$.

Av. length of crank end cards of water cyl. = 2.92"

$2.92 \times \frac{3.704}{12} = .901'$

$(901 + 899) \div 2 = 900'$.

3.704 = stroke ratio)

Item 5. Barometer

Barometer read 29.40" Hg.

2940 x .491 = 4.44 lbs. per sq. in.

Item 12. Temperature of saturated steam secured from steam table.

Item 1· Degrees Superheat.

Item 13 — Item 12

Item 18. Discharge Pressure (feet of water).

$$\frac{60 \times 144}{62.31} = 138.7$$

(62.31 = density of water at 70^{6})

Item 23. Steam per I. H. P. hour

$306 \div 1.584 = .01484$

Item 31. Specific Heat of Superheater steam secured from curves on page 137 in Professor Gebhardt's "Steam Power Plant Engineering."

Item 33. Cu. Ft. of water pumped

$$\frac{10307}{62.31} = 165.5$$

Item 34. Gallons of water pumped

$$\frac{165.5 \times 1728}{231} = 1238$$

Item 40. Plunger displacement per hr. (cu.ft.)

$A \times L \times N \times 60$

A = Av. piston area in sq. ft.

L = Length of stroke in ft.

N = Single strokes per min.

$$\frac{40.35}{144} \times 9 \times 22.17 \times 60 = 335.$$

Item 41.

Per Cent slip

$$\frac{\text{Item 40} - \text{Item 36}}{\text{Item 40}}$$

$$\frac{335-331}{335} = \frac{4}{335} = 1.19\%$$

Item 46. I. H. P. (steam)

$$\frac{PLAN}{33000}$$

$$\frac{23.3 \times 3.704 \left(\frac{.911 \times 905}{2}\right)\left(\frac{113.1+111.6}{2}\right)22.17}{33000 \times 12 \times 144} = 1.584$$

Item 47. I. H. P. (water)

$$\frac{59.6 \times .9 \times 4035 \times 22.17}{33000 \times 144} = 1.458$$

Item 48. D. H. P.

$$\frac{140.8 \times 20614}{33000 \times 60} = 1.468.$$

Item 49. Friction between cylinders (%).

$$\frac{\text{I. H. P. (Steam)} - \text{I. H. P. (Water)}}{\text{I. H. P. (Steam)}} =$$

$$\frac{1.584 - 1.458}{1.584} = 7.95$$

Item 50. Mech. Eff. = 100 -- 795 = 92.05%.

Item 51. B. T. U. Supplied per hour

$306\ (H+C_p\ S - q_2)$

$306\ [1167+(.58)(33.) - 211+32]$

$306\ (1007.) = 308,000.$

Item 52. B. T. U. per I. H. P. min.

$308,000 \div 60 \div 1.584 = 3240.$

Item 55. Duty per million B. T. U.

$\frac{140.8 \times 20614.}{308,000}\ 1000000 = 9,410,000$ ft. lbs.

Item 56. Termal Equiv. of work done per hour.

$1.468 \times 2545 = 3740$

Item 57. Thermodynamic Efficiency.

$\frac{3740 \times 100}{308,000} = 1.213\%$

Quality of Steam

$x_1\ r_1+q_1 = r_2+C_p\ (t_s - t_2)$

(Subscripts 1 correspond to conditions in the steam pipe and subscripts 2, to those in the calorimeter.)

P. = 46.93 lbs. Abs.

t_s = temperature of superheated steam at atmospheric pressure of 14.43 lbs.

C_p = specific heat of superheated steam in calorimeter. Its value was secured from curves on page 137 of Professor Gebhardt's "Steam Power Plant Engineering."

$$X_1 = (920.2)\ 245.6 = 1146.3\ .61(216.-211.1)$$

$$X_1 = \frac{1146.3\ 3. - 245.6}{920.2} = 98.24\%.$$